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(Reaffirmed 2001)

Indian Standard

METHODS OF EVALUATING DYNAMIC LOAD RATINGS OF ROLLING BEARINGS

PART 3 THRUST BALL BEARINGS

(First Revision)

1. Scope — Covers the methods for calculating the basic dynamic load ratings and rating life of thrust ball bearings.

1.1 The standard also specifies methods of calculation of adjusted rating life to take into account various reliabilities, materials and operating conditions.

1.2 This standard is not applicable to designs where the rolling elements operate directly on a shaft or housing surface, unless that surface is equivalent in all respects to the bearing rings (or washers) raceways it replaces.

2. Definitions

2.1 *Life* — For an individual rolling bearing, the number of revolutions which one of the bearing rings (or washers) makes in relation to the other rings (or washers) under the prevailing working conditions before the first evidence of fatigue develops in the material of one of the rings (or washers) or rolling elements.

2.2 *Reliability* (*In the context of bearing life*) — For a group of apparently identical rolling bearings, operating under the same conditions, the percentage of the group that is expected to attain or exceed a specified life.

2.2.1 The reliability of an individual rolling bearing is the probability that the bearing will attain or exceed a specified life.

2.3 *Basic Rating Life* — For an individual rolling bearing, or a group of apparently identical rolling bearings operating under the same conditions, the life associated with 90 percent reliability.

2.4 *Basic Dynamic Radial Load Rating* — That constant stationary radial load which a rolling bearing can theoretically endure for a basic rating life of one million revolutions. In the case of a single row angular contact bearing, the radial load rating refers to the radial component of that load which causes a purely radial displacement of the bearing rings in relation to each other.

2.5 *Basic Dynamic Axial Load Rating* — That constant centric axial load which a rolling bearing can theoretically endure for a basic rating life of one million revolutions.

2.6 *Dynamic Equivalent Radial Load* — That constant stationary radial load under the influence of which a rolling bearing would have the same life as it will attain under the actual load conditions.

2.7 *Dynamic Equivalent Axial Load* — That constant centric axial load under the influence of which a rolling bearing would have the same life as it will attain under the actual load conditions.

2.8 *Nominal Contact Angle* — The angle between a plane perpendicular to the bearing axis and the nominal line of action of the resultant of the forces transmitted by a bearing ring to a rolling element.

Rolling Bearings Sectional Committee, EDC 39 [Ref : Doc : EDC 39 (3638)]

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BUREAU OF INDIAN STANDARDS
MANAK BHAVAN, 9 BAHADUR SHAH ZAFAR MARG
NEW DELHI 110002

3. Symbols — For the purpose of this standard, the following symbols shall apply:

C_r = Basic dynamic radial load rating, newtons

C_a = Basic dynamic axial load rating, newtons

C_{or} = Basic static radial load rating, newtons

C_{oa} = Basic static axial load rating, newtons

D_w = Ball diameter, millimetres

D_{pw} = Pitch diameter of ball, millimetres

F_r = Bearing radial load = radial component of actual bearing load, newtons

F_a = Bearing axial load = axial component of actual bearing load, newtons

L_{10} = Basic rating life, million revolutions

L_n = Adjusted rating life for a reliability of ($100-n$) percent, million revolutions where n = Adjusted rating life

L_{10a} = Adjusted rating life for non-conventional material and operating conditions, million revolutions

L_{na} = Adjusted rating life for non-conventional material and operating conditions and for a reliability of ($100-n$) percent, million revolutions

P_r = Dynamic equivalent radial load, newtons

P_a = Dynamic equivalent axial load, newtons

X = Radial load factor

Y = Axial load factor

Z = Number of balls in a single row bearing, number per row of a multi-row bearing with equal number of balls per row

a_1 = Life adjustment factor for a reliability other than 90 percent

a_2 = Life adjustment factor for non-conventional material

a_3 = Life adjustment factor for non-conventional operating conditions

e = Limit value of F_a/F_r for the applicability of different values of factors X and Y

f_c = A factor which depends on the geometry of the bearing components, the accuracy to which the various components are made and the material

i = Number of rows of balls in a bearing

α = Nominal contact angle of a bearing, degrees

4. Basic Dynamic Axial Load Rating

4.1 *Single Row Bearings* — The basic dynamic axial load rating for single row, single or double direction thrust ball bearing shall be calculated as follows:

For $D_w \leqslant 25.4$ mm $\alpha = 90°$:

$$C_a = f_c\, Z^{2/3} D_w^{1.8}$$

For $D_w \leqslant 25.4$ mm $\alpha \neq 90°$:

$$C_a = f_c\, (\cos\alpha)^{0.7} \tan\alpha\, Z^{2/3} D_w^{1.8}$$

For $D_w > 25.4$ mm $\alpha = 90°$:

$$C_a = 3.647\, f_c\, Z^{2/3} D_w^{1.4}$$

For $D_w > 25.4$ mm $\alpha \neq 90°$:

$$C_a = 3.647\, f_c\, (\cos\alpha)^{0.7} \tan\alpha\, Z^{2/3} D_w^{1.4}$$

Z is the number of balls carrying load in one direction.

4.1.1 Values of factor f_c shall be as given in Table 1 and apply to bearings with a cross-sectional raceway groove radius not larger than 0·54 D_w. The load carrying ability of a bearing is not necessarily increased by the use of a smaller groove radius, but is reduced by the use of a larger groove radius than that indicated above.

TABLE 1 VALUES OF FACTOR f_c FOR THRUST BALL BEARINGS

$\frac{D_w}{D_{pw}}$	f_c $\alpha = 90°$	$\frac{D_w \cos \alpha}{D_{pw}}$	f_c $\alpha = 45°$	$\alpha = 60°$	$\alpha = 75°$
0·01	36·7	0·01	42·1	39·2	37·3
0·02	45·2	0·02	51·7	49·1	45·9
0·03	51·1	0·03	58·2	54·2	51·7
0·04	55·7	0·04	63·3	58·9	56·1
0·05	59·5	0·05	67·3	62·6	59·7
0·06	62·9	0·06	70·7	65·8	62·7
0·07	65·8	0·07	73·5	68·4	65·2
0·08	68·5	0·08	75·9	70·7	67·3
0·09	71·0	0·09	78·0	72·6	69·2
0·10	73·3	0·10	79·7	74·2	70·7
0·12	77·4	0·12	82·3	76·6	—
0·14	81·1	0·14	84·1	78·3	—
0·16	84·4	0·16	85·1	79·2	—
0·18	87·4	0·18	85·5	79·6	—
0·20	90·2	0·20	85·4	79·5	—
0·22	92·8	0·22	84·9	—	—
0·24	95·3	0·24	84·0	—	—
0·26	97·6	0·26	82·8	—	—
0·28	99·8	0·28	81·3	—	—
0·30	101·9	0·30	79·6	—	—
0·32	103·9	—	—	—	—
0·34	105·8	—	—	—	—

Note — For thrust bearings α>45°, values for α=45° are shown to permit interpolation of values for α between 45° and 60°.

4.2 *Bearings with Two or More Rows of Balls* — The basic dynamic axial load rating for thrust ball bearings with two or more rows of similar balls carrying load in the same direction shall be calculated as:

$$C_a = (Z_1 + Z_2 + \dots + Z_n) \times \left[\left(\frac{Z_1}{C_{a1}}\right)^{10/3} + \left(\frac{Z_2}{C_{a2}}\right)^{10/3} + \dots + \left(\frac{Z_n}{C_{an}}\right)^{10/3}\right]^{-3/10}$$

The load ratings C_{a1}, C_{a2}, , C_{an} for the rows with Z_1, Z_2, ... , Z_n balls are calculated from the appropriate single row bearing formula (*see* **4.1**). Values of f_c for $\frac{D_w}{D_{pw}}$ or $\frac{D_w \cos \alpha}{D_{pw}}$ and/or contact angles other than shown in the table are obtained by linear interpolation or extrapolation.

5. Dynamic Equivalent Axial Load — The equivalent axial load for thrust ball bearings with $\alpha \neq 90°$, under constant radial and axial loads shall be given as:

$$P_a = X F_r + Y F_a$$

Values of factors X and Y shall be given as in Table 2.

Thrust ball bearings with $\alpha = 90°$, can support axial loads only. The equivalent load for this type of bearing is:

$$P_a = F_a$$

TABLE 2 VALUES OF FACTORS *X* AND *Y* FOR THRUST BALL BEARINGS

(*Clause* 5)

α	Single Direction Bearings*		Double Direction Bearings				e
	$\frac{F_a}{F_r} > e$		$\frac{F_a}{F_r} < e$		$\frac{F_a}{F_r} > e$		
	X	Y	X	Y	X	Y	
45°	0·66		1·18	0·59	0·66		1·25
50°	0·73		1·37	0·57	0·73		1·49
55°	0·81		1·60	0·56	0·81		1·79
60°	0·92		1·90	0·55	0·92		2·17
65°	1·06	1	2·30	0·54	1·06	1	2·68
70°	1·28		2·90	0·53	1·28		3·43
75°	1·66		3·89	0·52	1·66		4·67
80°	2·43		5·86	0·52	2·43		7·09
85°	4·80		11·75	0·51	4·80		14·29
$\alpha \neq 90°$	$1{\cdot}25 \tan\alpha \left(1 - \frac{2}{3}\sin\alpha\right)$	1	$\frac{20}{13}\tan\alpha\left(1 - \frac{1}{3}\sin\alpha\right)$	$\frac{10}{13}\left(1 - \frac{1}{3}\sin\alpha\right)$	$1{\cdot}25 \tan\alpha \left(1 - \frac{2}{3}\sin\alpha\right)$	1	$1{\cdot}25 \tan\alpha$

Note — For thrust bearings $\alpha > 45°$. Values for $\alpha = 45°$ are shown to permit interpolation of values for α between 45° and 50°.

*$\frac{F_a}{F_r} < e$ is unsuitable for single direction bearings.

6. Basic Rating Life — The basic rating life for thrust ball bearings shall be given as:

$$L_{10} = \left(\frac{C_a}{P_a}\right)^3$$

The values of C_a and P_a are calculated in accordance with **4.1** and **4.2**.

This life formula gives satisfactory results for a broad range of bearing loads. However, extra heavy loads may cause detrimental plastic deformations at the ball/raceway contacts. The user shall, therefore, consult the bearing manufacturer to establish the applicability of the life formula in cases where P_a exceeds 0·5 C_a.

7. Adjusted Rating Life

7.1 *General*

7.1.1 *Reliability level* — The normal criterion of bearing performance is the basic rating life calculated according to this standard and this life is associated with 90 percent reliability. However, for certain applications it may be desirable to calculate the life for other reliability levels.

The adjusted rating life for a reliability of (100 — n) percent shall be:

$$L_n = a_1 L_{10}$$

Values of factor a_1 are given in **7.2**.

7.1.2 *Material and operating conditions* — It is recognized that the properties of the material and the operating conditions have an influence on bearing life. The basic rating life calculated according to this standard is associated with conventional material (good quality, hardened steel) and conventional operating conditions (a bearing properly mounted, adequately lubricated, protected from foreign matter, conventionally loaded and not exposed to extreme temperature).

In certain cases the bearing material characteristics and/or operating conditions deviate from the conventional in such a way that it is justified to take their influence into special consideration.

The adjusted rating life for non-conventional materials and operating conditions shall be given as:

$$L_{10a} = a_2 a_3 L_{10}$$

The adjusted rating life for non-conventional materials and operating conditions and for a reliability of (100 — n) percent shall be given as:

$$L_{na} = a_1 a_2 a_3 L_{10}$$

Values of factors a_2 and a_3 are discussed in 7.3 and 7.4.

7.1.3 *Limitations* — In addition to the required fatigue life, other factors such as maximum permissible bearing deflection and minimum shaft and housing strength, shall be given due consideration when selecting the size of bearings for a given application. Particular discretion shall be exercised when utilizing adjusted rating life values which are greater than L_{10}.

7.2 *Life Adjustment Factor for Reliability* — The adjusted rating life for a reliability of (100 — n) percent shall be calculated in accordance with 7.1.1. Values of the pertinent adjustment factor a_1 shall be as given in Table 3.

TABLE 3 VALUES FOR LIFE ADJUSTMENT FACTOR FOR RELIABILITY, a_1

Reliability Percent	L_n	a_1
90	L_{10}	1
95	L_5	0·62
96	L_4	0·53
97	L_3	0·44
98	L_2	0·33
99	L_1	0·21

7.3 *Life Adjustment Factor for Material* — Currently the selection of a_2 values cannot be based on quantifiable material characteristics but only on test results and other experience made with bearings. Values of a_2 shall, therefore, be obtained from the bearing manufacturer.

The use of a certain steel analysis and/or process as such is not sufficient justification for the use of an a_2 value other than 1. Values of a_2 greater than 1 may, however, be applicable to bearings made of steel of particularly low impurity content or of special analysis.

Bearing manufacturing processes affecting the properties of the material in a bearing are also to be considered in the selection of an a_2 value. If, for example, a reduced life is expected because of a hardness reduction caused by special heat treatment, this should consequently be considered by the manufacturer's selection of a correspondingly reduced a_2 value.

It may not be assumed that the use of an improved material will overcome a deficiency in lubrication. Values of a_2 greater than 1 should, therefore, normally not be applied where factor a_3 is less than 1 because of such deficiency.

7.4 *Life Adjustment Factor for Operating Conditions* — Of the operating conditions directly influencing bearing life, the magnitude and direction of the load are considered in the calculation of the equivalent load (*see* **5** of Part 1) and deviations from normal load distribution are discussed in the explanatory note.

Operating conditions which remain to be taken into account here are the adequacy of the lubrication (at the operating speed and temperature) and conditions causing changes in material properties (for example high temperature causing reduced hardness). The influence on bearing life of such conditions may be considered by the application of a life adjustment factor a_3.

The calculation of basic dynamic load rating and basic rating life in this standard assumes that bearing life is limited principally by sub-surface fatigue, that is, that the rolling elements and the rings (washers) raceways are sufficiently separated by a lubricant to make the probability of failures caused by surface distress negligible. Where this requirement is fulfilled $a_3 = 1$ provided a lower value does not apply, for example because of a change in material properties caused by the operating conditions.

Reduction of a_3 values should be considered, for example, where the viscosity of the lubricant is less than 13 mm²/s (1 mm²/s = cSt) for ball bearings at the operating temperature where the rotational speed is exceptionally low (revolutions per minute multiplied by D_{pw} less than 10 000). Values of a_3 greater than 1 may be considered only where the lubrication conditions are particularly favourable.

It may not be assumed that a deficiency in lubrication can be overcome by using an improved material. Where factor a_3 is less than 1, due to inadequate lubrication, values of a_3 greater than 1 should, therefore, normally not be used.

EXPLANATORY NOTE

This standard was first published in 1966. This standard is being revised to align it with ISO 281/1-1977, 'Rolling bearings — Dynamic load ratings and rating life — Part 1 Calculation methods'. In this revision, adjusted rating life of bearings has been included.

This standard consists of the following parts:

IS : 3824 (Part 1)-1983 'Methods of evaluating dynamic load ratings of rolling bearings: Part 1 Radial ball bearings (*first revision*)'

IS : 3824 (Part 2)-1983 'Methods of evaluating dynamic load ratings of rolling bearings: Part 2 Radial roller bearings (*first revision*)'

IS : 3824 (Part 3)-1983 'Methods of evaluating dynamic load ratings of rolling bearings: Part 3 Thrust ball bearings (*first revision*)'

IS : 3824 (Part 4)-1983 'Methods of evaluating dynamic load ratings of rolling bearings: Part 4 Thrust roller bearings (*first revision*)'

Ball and roller bearings, collectively known as rolling bearings are being used in all modern machines. This standard is intended to help manufacturers in the proper design of these bearings.

It is often impractical to establish the suitability of a bearing related for a specific application by testing a sufficient number of bearings in that application. Other methods are, therefore, required to establish this suitability.

A reliable life calculation is considered to be a suitable and advantageous substitute for testing.

Calculations according to this standard do not yield satisfactory results for bearings subjected to such application condition and/or of such internal design which results in considerable truncation of the area of contact between the rolling elements and the ring raceways. Unmodified calculation results are thus not applicable, for example, to groove ball bearings with filling slots which project substantially into a ball/raceway contact area when the bearing is subjected to load in application.

Calculations according to this standard do not yield satisfactory results for bearings subjected to such application conditions which cause deviations from a normal load distribution in the bearing, for example, misalignment, having shaft deflection, rolling element centrifugal forces or other high speed effects, and preload or extra large clearance in radial bearings. Where there is reason to assume that such conditions prevail, the user should consult the bearing manufacturer for recommendations and evaluation of equivalent load and life.

Reprography Unit, BIS, New Delhi, India